Design Project Final Report

Designing, analyzing, and assembling a manually actuated worm gear system to rotate a 10-pound weight on a one-foot moment arm 180º, within a 24”x12”x6” envelope and a \$125 budget.

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ME 329 - 02

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RE: Design Project: Final Report

Abstract

This project involved designing and building a manually actuated gearbox to rotate a 10-pound weight on a one-foot moment arm through 180º of rotation within a 24"x12"x6" envelope and a $125 budget.

A worm gear system was selected immediately for its high reduction ratio and self-locking capabilities. In order to stay within budget, off the shelf components and a wood frame were used. Critical components were analyzed using AGMA standards and mechanical design references such as Shigley's Mechanical Engineering Design. A SolidWorks model with motion and numerical analysis verified overall function and structural performance. Testing revealed gear misalignment and snap ring failure, but the project met core design goals and provided insights for future improvements.

Design Justification

The initial task for this design challenge was to create a non-electric system that can flip 10 pounds of weight on a one-foot moment arm by 180 degrees to simulate the action of a Powder Bed Coating suppository that can deliver metallic powder to multiple beds efficiently. The initial design is set to support 120 in-lb of torque. This system requires a spatial constraint of 24"x12"x6". Upon request of the client, the system must be manually actuated with an input force of roughly 10 pounds at 30-60 RPM. The device must mitigate back driving, a well-known feature of worm assemblies, and a preferred (but not required) 90 degree turn from input to output. Finally, the cost of the design and manufacturing must remain within $125 dollars.

Worm gears provide high, single stage reduction, removing complexity, as well as self-locking features. Additionally, low efficiency was not a concern in this application, as no power constraint was provided. As a result of extensive research into different worm gear systems, it became clear that the average price range of an Imperial dimensioned worm gear assembly is well outside the given budget. Similar budgeting issues arose while looking into bevel and helix gear systems as well. Ultimately, a metric worm gear with a 12 mm shaft diameter was chosen, as metric gears are noticeably cheaper. By drilling the shaft hole to fit the given ½ inch (12.7 mm) shaft, the budget can be maintained, and torque transmission will not be sacrificed. Plain bearings were chosen to constrain the shafts for smooth operation and low cost. These were mounted with a slip fit into the frame on either side of the housing, with the flange on the external side. This configuration, with the flange on the outside, was chosen for ease of assembly, as putting c-clips with the shafts inside the housing would be time consuming and have a comparable success rate.

The shafts were axially constrained with said c-clips and set into grooves on the shaft. This was the chosen method of axial constraint for its easy assembly, low cost, and theoretical capability to withstand the necessary axial loads. For rotationally constraining the gears, set screws seemed risky, as they are liable to slip under the relatively heavy loads that were expected for this application. Keyways provide higher rotational capacity but are much more difficult to manufacture, leading to the decision to fix the gears to the shaft using much cheaper metal dowel pins. This method of constraint was much simpler to manufacture, only requiring one hole drilled through each shaft and gear collar. Additionally, it provides easy locating along the shaft, which set screws or keyways do not. The only notable downside of pin is that once assembled, disassembly is very difficult due to the transition fit of the pin.

For the frame, metal was clearly the strongest, most reliable option, most likely aluminum sheet. However, this extra strong material would increase costs and jeopardize budget constraints. To compromise between strength and cost, wood was used. The most robust wood option available was a 2"x8" pine slab from Home Depot, which was used to assemble the frame with 12 screws which were configured to withstand the maximum axial loading.

Numerical Analysis

To ensure the reliability and performance of the gearbox design, a comprehensive numerical analysis of all critical components was conducted, including the driving gear, shaft, bearings, and pin. Using *Shigley's Mechanical Engineering Design* and AGMA standards as reference, calculated stresses, safety factors, and wear limits were analytically confirmed, and mechanical and durability requirements were met.

Gears

In accordance with *Shigley's Mechanical Engineering Design* and AGMA standards, our analysis focused solely on the driving gear in our worn gear system. This approach is standard practice because the driving gear typically experiences higher bending and contact stress due to its geometry, material properties, and the nature of sliding contact. The worm, often made of harder material and shaped like a screw is less prone to failure and not considered the limiting component in most designs. AGMA formulas and geometry factors are primarily developed for the gear, where failure is more likely to initiate. As such, analyzing only the worm gear provides a conservative and effective assessment of the system's strength and performance.

Gear	Pitch (mm)	Diameter (mm)	Teeth	RPM	Torque (hp)
Worm	32	28.575	1	30	
Gear	32	19.05	20	60	4.27 E -7

Table 1. Gear Parameters

Bending/Allowable Stress (psi)	Bending Safety Factor	Contact/Allowable Stress (psi)	Contact Safety Factor
4195.7	4.507	206492.25	0.42

Table 2. Driving Gear Analysis

Shaft

Diameter (in)	Yield Strength (ksi)	Ultimate Strength (ksi)	Kt	Kts	Kf	Kfs
0.5	84	144	5	3	5	3

Table 3. Shaft Parameters

Max Transverse Load (lb)	Max Torque (ft*lb)	Min Transverse Load (lb)	Min Torque (ft*lb)	Load Distance to Bearings	Shaft Length
100	10	0	0	6	12

Table 4. Shaft Input Parameters

Ka	Kb	Kc	Kd	Ke
1	1.211329	1	1	0.753

Table 5. Shaft Fatigue Factors

Tau Max (ksi)	Tau Alternate (ksi)	Von Mises Stress (ksi)	Von Mises Safety Factor	Goodman Safety Factor
24.4	12.2	21.17	3.97	3.75

Table 6. Shaft Analysis

Bearing

Radial Load	Diameter	Bushing Lengths	RPM	Cycles
100	0.5	0.75	60	55000

Table 7. Bearing Parameters

Pressure	Pressure Max	Surface Speed	Surface Distance (ft)
266.67	339.53	7.85	7199.27

Table 8. Bearing Pressure and Speed

Wear Factor	Wear (in)	Allowed Wear	Safety Factor
3*10^-10	0.000733312	0.0016	2.18

Table 9. Bearing Wear

Allowed PV	PV	Safety Factor
18000	2666.67	6.75

Table 10. Bearing Pressure Velocity

Pin

Torque (in*lb)	**Shaft Diameter** (in)	**Pin Diameter** (in)	**Pin Ultimate Strength** (ksi)
120	0.5	0.125	70

Table 11. Pin Parameters

Shear Load	Shear Strength	Safety Factor
480	1718.06	3.58

Table 12. Pin Analysis

SolidWorks

To support and visualize the gearbox design, a CAD model of the entire assembly was built using SolidWorks. This included a full assembly drawing with detailed components of placement and fitment, as well as a complete Bill of Materials to document all purchased and manufactured parts. Beyond the static model, SolidWorks Simulation tools were utilized to evaluate the mechanical performance of the overall design. A Finite Element Analysis was conducted to assess stress concentrations and deformation under expected loading conditions, ensuring structural integrity. Additionally, a motion simulation was performed to visualize the dynamic behavior of the gearbox and confirm that the input torque and gear interactions would successfully generate the desired output torque. Together, these tools helped to verify the functionality, identify potential design issues, and refine the assembly before manufacturing.

Motion Simulation

The CAD assembly motion study verifies that 10 rotations of the input shaft equate to 1 half rotation at the output, which is expected by the chosen gear ratio. Clearance for gears at every point in rotation is also ensured, as well as at least 6 inches between the input shaft axis and weights.

Assembly Drawing and Bill of Materials

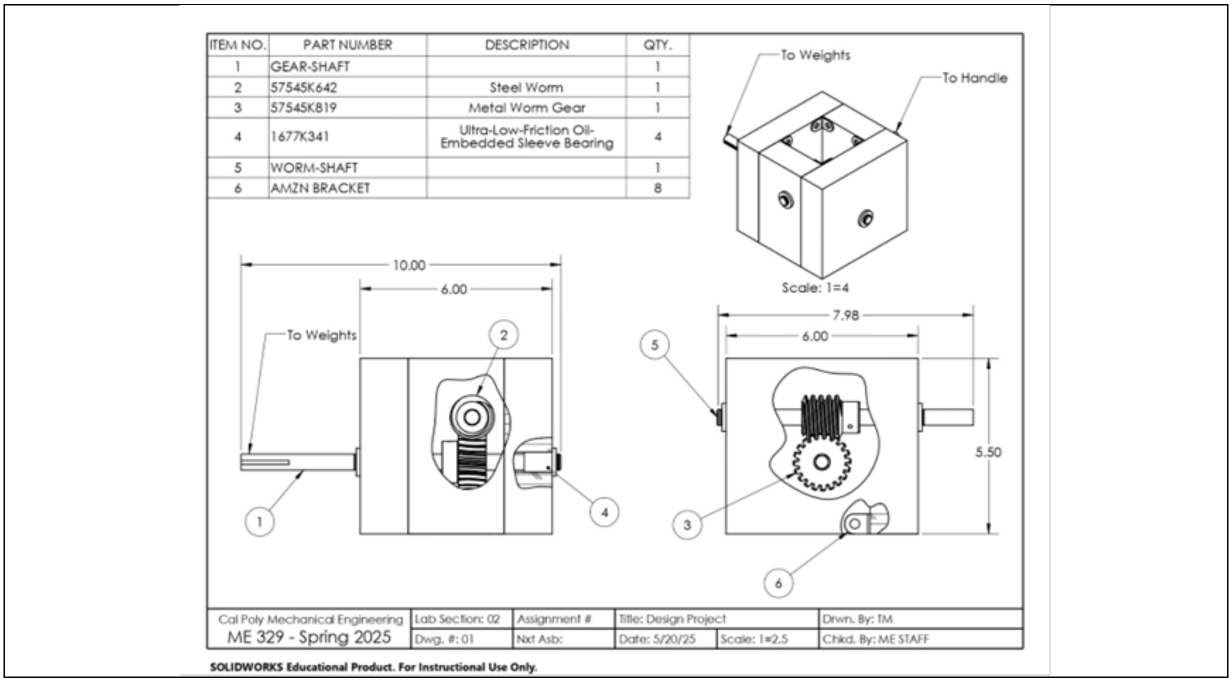


Figure 1. Assembly Drawing

Vendor	Part Number	Length	Quantity	Total Cost	Description
McMaster-Carr	57545K642		1	$27.28	Metric 1045 Carbon Steel Worm, Module 2
McMaster-Carr	57545K819		1	$28.47	Metric Cast Iron Worm Gear, Module 2
McMaster-Carr	1677K341		4	$12.80	Low-Friction Oil-Embedded Flanged Sleeve Bearing
McMaster-Carr	1346K512	18 in	1	$20.52	Rotary Shaft, 1566 Carbon Steel, 1/2" Diameter, 18" Long
Home Depot	329444946	8 ft	1	$9.56	2" x 8" x 8' Fir Board
McMaster-Carr	98381A494	1.375 in	1	$3.93	Pack of 5 Dowel Pin, Alloy Steel, 1/8" Diameter, 1-3/8" Long
Home Depot	204645668	2.5 in	8	$7.20	1/2 in. x 2-1/2 in. Hex Zinc Plated Lag Screw

Sum	$109.76
Total Including Tax	$118.54

Table 13. Bill of Materials

Manufacturing and Assembly

Manufacturing for this assembly was significantly slowed by Cal Poly Senior Project, who required significant time and resources within shop spaces. In particular, lathe and mill availability were sparse for the entire month leading up to the last week of Spring classes. While this did not prevent progress on all manufacturing tasks, it did limit available time for testing and create a sense of rushed work time.

Manufacturing tasks varied in difficulty. Tasks such as drilling out gearboxes and pin holes were relatively quick and easy tasks, and the drill presses needed to complete them were readily available. Parting our shaft into two sections and milling keyways were slightly more difficult operations. The chosen shaft was hardened, which required the abrasive saw to cut. This saw leaves a rough surface finish, which needed post processing to clean up. The milling of the keyways was time-consuming, as mills were frequently in use, and the mills that were available were not in the correct configuration, or in working order at all. Additionally, milling takes more skill and setup time than drilling.

The most difficult task of the manufacturing process was the cutting of the circlip grooves. On top of lathes being difficult to access due to senior project, the hardened shaft was difficult to cut, the tooling that was available was the incorrect geometry and chipped, and the lathes that were available had significant issues with chuck runout. All these factors led to grooves that were too wide and had inconsistent depth, leading to one of the clips popping off during final testing. In future iterations, more time will be allotted for this task, as time constraints led to the use of improper tooling.

Figure 2. Fully Assembled Gearbox

Assembly was relatively simple, but did encounter some problems. Installing the gears to the shaft was simple. The gears were slid on, the pin holes were aligned, and the dowel pins were driven with an arbor press. The bearings were fitted into the wood frame pieces, and the frame pieces slid onto the shafts. The c-clips were then installed to retain the shafts, as seen in figure 2. The main assembly setback arose when finally screwing the frame pieces together. While driving the screws, the clamping force of the screw itself pulled the assembly out of alignment, causing binding, or the gears to not engage at all. The worm and worm gear required precise radial spacing to function properly, and the simple construction of our wood frame would not allow for that, leading to the gear teeth slipping during testing.

Design Iterations and Testing

Throughout the project, the design process remained relatively stable, with only a few modifications made between the initial concept and final assembly. One of the earliest planned features was the use of the L-Bracket to reinforce the internal corners or the wooden gearbox frame. However, as the design progressed, it was determined that these were unnecessary, given the sturdiness of the wood and the simplicity of the load path. Removing the L-Brackets reduced both the cost and assembly time without significantly compromising structural integrity.

After the full assembly of the gearbox, functional tests were conducted beginning with a motor actuation test, where the gearbox was manually rotated to confirm proper meshing between the worm and its driving gear. This test showed that the gears were properly aligned and operated smoothly with minimal backlash. The shaft and bearings held their positions securely during these rotations. The wooden frame provided sufficient rigidity for the system, though for prolonged use, switching to a metal frame to improve structural stiffness would be ideal. The testing process confirmed that the design meets the basic functional requirements. It transmits motion with the required speed reduction and torque increase, and all mechanical connections, performed as expected during initial use.

The final test used a 10 lb weight positioned 11 inches from the output shaft. Unfortunately, the test revealed multiple critical manufacturing errors, including improper fitting between the worm and driving gear as seen in figures 3 and 4. Due to a manufacturing error, the gears were spaced further apart than initially planned, and the small about of contact between gears resulted in skipping under the high load. Additionally, a snap ring used to axially constrain the shaft failed during operation. Without the appropriate tools on hand to reinstall or replace the snap ring, continued testing was impossible.

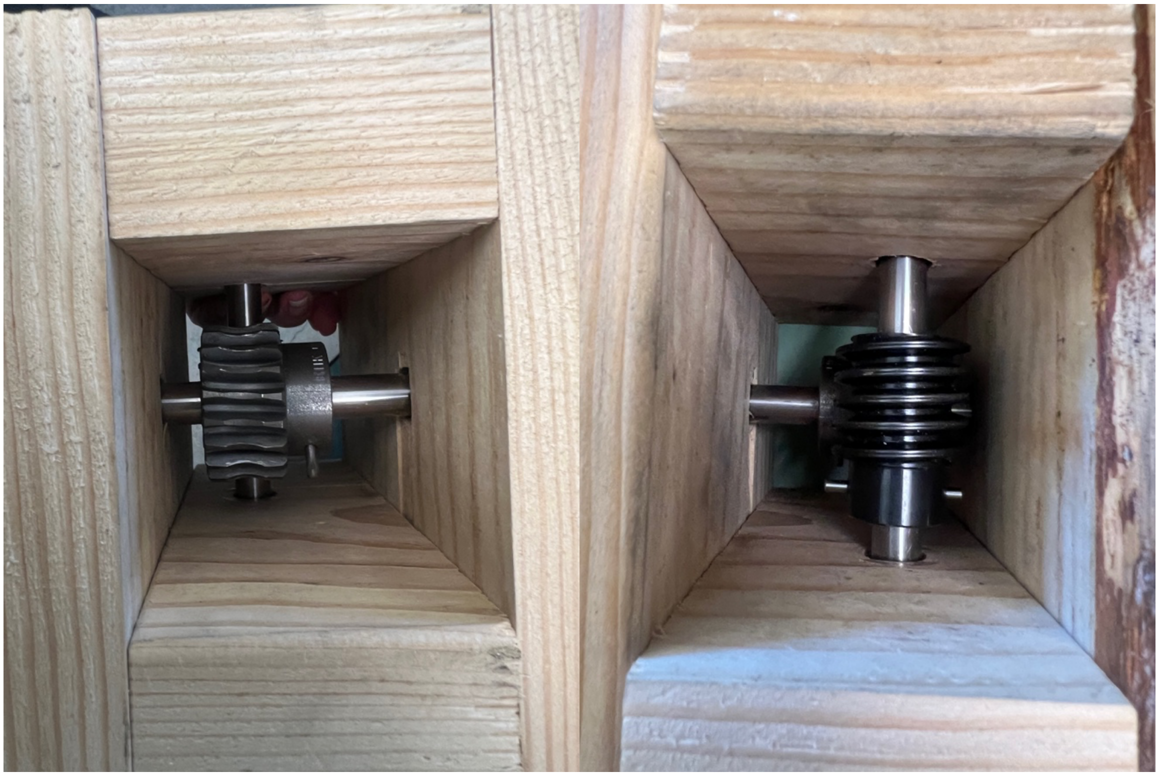


Figure 3 and 4. Internal View of Gearbox

Despite these setbacks, the test provided useful feedback on gear alignment and shaft retention that would guide future improvement in this design.

Expected Modifications

In future iterations, several changes could be expected to improve the overall functionality and reliability of the powder bed coating device. The primary point of failure during testing was the wooden housing, which most likely required more axial reinforcement to prevent the box from shearing itself apart during operation of the worm train. The housing was also dimensioned in a way that caused interference between the handle and the moment arm of the weighted powder suppository. To fix this, a stronger housing material such as steel or aluminum plating with welded joints or precision bolt holes would be used to support the inner design, and the handle would be positioned farther back along the housing to give adequate spacing between the lever arm and the weight bearing shaft. An alternative for this solution would be a shorter handle, but a longer handle provides more leverage for the user, who might prefer less input power when operating the device several times a day. By introducing precision bolt holes to join the sides of the housing, movement of shaft placement and overall alignment issues would be essentially eliminated, also aiding in the smoothness of the gear interface.

The root cause for the ultimate disassembly of the device during testing was inadequate spacing between the worm and worm gear. The interface between gear teeth was too small, meaning the teeth were not properly enmeshed. This caused unintentional locking in both directions and slippage when too much torque was applied. By installing the shafts carrying each gear closer radially by about .075 in, the smoothness and reliability of the train would dramatically improve. This solution would mainly be addressed during manufacturing, where the precision of the shaft placement withing the housing walls must be increased.

To improve the fundamental design of this assembly, changing materials and improving precision during manufacturing will drastically improve the performance of the device, allowing the mechanical components to operate to their full potential.

Conclusion

Despite limited resources and manufacturing challenges, the design, manufacturing, and assembly of a functional gearbox, capable of delivering the required torque to flip a 10-pound weight, was a success. The use of a worm gear provided the necessary mechanical advantage and self-locking behaviors within the compact size and budget constraints. While testing revealed issues with gear alignment and snap ring retentions, these highlighted key areas for improvement in manufacturing precision and shaft support. Overall, the project met its primary objectives, and the design process provided valuable experience in mechanical system design, analysis, and assembly under real-world constraints.

Appendix

[A] Gear Calculations

[B] Shaft Calculations

[C] Bearing Calculations

[D] Pin Calculations

[E] Assembly Drawing and Bill of Material

[F] Worm Shaft Drawing

[G] Gear Shaft Drawing

[H] Worm Shaft Drawing

[I] Gear Drawing

[J] Frame Drawing

Gear Parameters					
	Pitch (in)	Diameter	Teeth	rpm	Torque (in-lb)
A (Worm)	1.26	1.125	1	3	
B (Gear)	1.26	0.750	20	60	1.20E+02
	32 mm				

Torque of Design		
Wt (worm)	1.34E+06	lb
Efficiency	0.6	%
Tout	1.20E+02	lb-in
Wt (gear)	3.20E+02	lb

AGMA Analysis Parameters			
Worm	**Life Span**	20	years
	1 Cycle	10	cycle/day
		260	day/yr
		5.20E+04	cycles
Gear	**Life Span**		
		5.20E+04	cycles

Kw= 22 (High-Tested Cast Iron/Cast Steel, 14.50º)

Gear/Worm Parameters			
Material = Steel			
F of Gear	0.866142	in	cast iron
F of Worm		in	1045 carbon steel
St Gear	15,954	psi	110MPa
Sc Gear	55,000	psi	

Bending Stress of Gear			
Wt	320		
K0	1.75		Moderate Shock for driving
Pitch-Line Velocity	3.92699082	ft/min	
Qv	8		Average gear quality
A	70.5588154		
B	0.6328783		
Kv	1.01768408		
Y	0.337		
Ks	1.13483193		
Pd	1.260	in	
F	0.866142	in	
Cmc	1		Uncrowned Teeth
Cpf	-0.0125		
Cma	0.14		
Ce	1		Normal Assembly
Cpm	1		Normal Mounted Pinion
A	0.127		Commercial, Enclosed
B	0.0158		Commercial, Enclosed
C	-9.30E-05		Commercial, Enclosed
Km	1.13		
Kb	1		No change in rim thickness
mn	1		
J	0.25338346		
Bending/Allowable Stress	4188.79611		psi
Yn	1.19		
Kt	1		temp below 250F
Kr	1		
Safety Factor	4.514		

Contact Stress of Gear			
Cp	2100		Steel driving Cast Iron
Cf	1		No additional effects
mg	24		
I	0.116354309		
Ch	1.180992661		
Contact/Allowable Stress	206492.25	psi	
Zn	1.34E+00		
Safety Factor	0.42		

$$\sigma_H = C_p\sqrt{\frac{W_t K_o K_v K_s K_m K_B}{d_p F I}}$$

$$\sigma = \frac{W_t K_o K_v K_s \cdot P_d K_m K_b}{F J}$$

Shaft Parameters						
Diameter (in)	Yield Strength (ksi)	Sut (ksi)	Kt	Kts	Kf	Kfs
0.5	84	144	5	3	5	3

Fatigue Factors				
Ka	Kb	Kc	Kd	Ke
1	1.211329	1	1	0.753

Input Parameters					
Transverse Load (lb	Torque (Ft*lb Max)	Transverse Load (Min)	Torque (Min)	Load Distance to Bearing	Shaft length (In, bearing to bearing)
100	10	0	0	6	12

Output											
Moment (in*lb)	SigmaMax (ksi)	TauMax (ksi)	SigmaAlternate	TauAlternate			Se	A	B	Von Mises	
0	**0**	**24.44621991**	**0**	**12.22310995**			**65.67342**	**6351.314**	**0**	**21.17105**	

Radial Bearing Loa	Radial Bearing Load 2	Von Mises Safety	Goodman Safety	Diameter (in)	Deflection
50	**50**	**3.967682758**	**0.253784791**		

infinite life

finite life based on 5.2*10^4 cycles

3.754363674

BEARINGS

Radial Load	Diameter	Bushing Length	Pressure
100	0.5	0.75	**266.6666667**

RPM	Surface Speed
60	**7.853975**

PressureMax	Cycles	Surface Distance (ft)
339.5308321	55000	**7199.270833**

Wear Factor	Wear (in)	Allowed Wear	Safety Factor
3E-10	**0.000733312**	0.0016	**2.181880688**

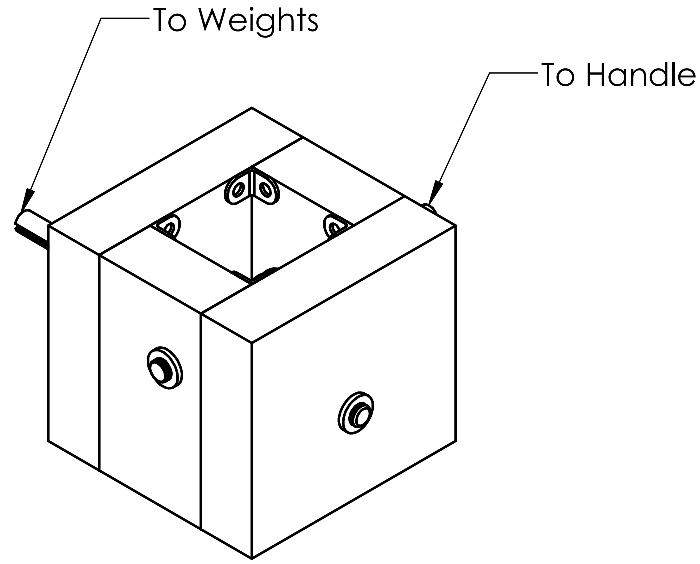
Allowed PV	PV	Safety Factor
18000	**2666.666667**	**6.75**

Pin

Torque (in*lb)	Shaft Diameter (in)	Pin Diameter (in)	Pin Sut (ksi)
120	0.5	0.125	70

Shear Load	Shear Strength	Safety Factor
480	**1718.057031**	**3.579285482**

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	GEAR-SHAFT		1
2	57545K642	Steel Worm	1
3	57545K819	Metal Worm Gear	1
4	1677K341	Ultra-Low-Friction Oil-Embedded Sleeve Bearing	4
5	WORM-SHAFT		1
6	AMZN BRACKET		8



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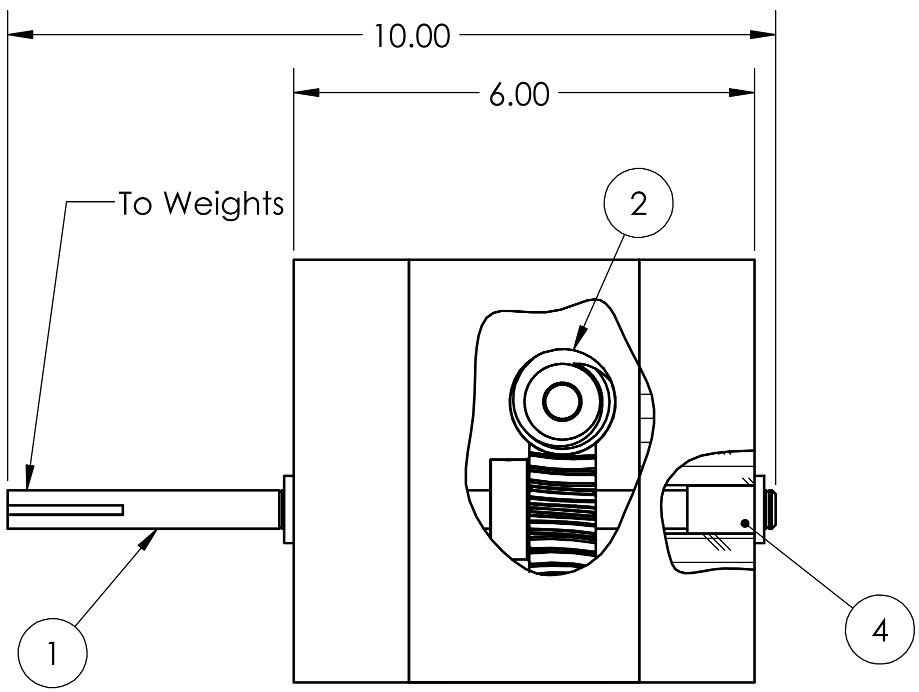

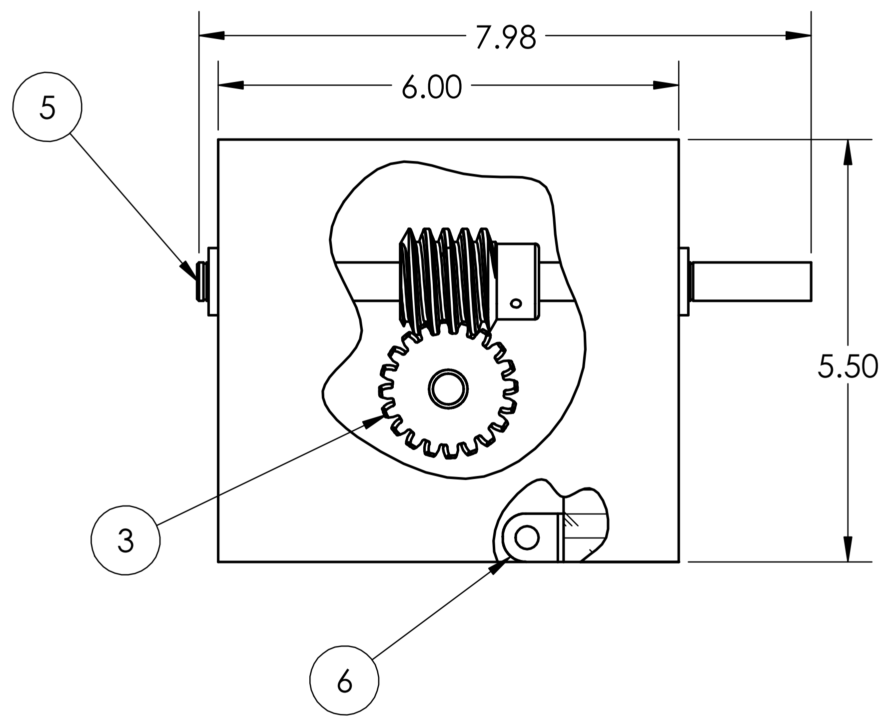

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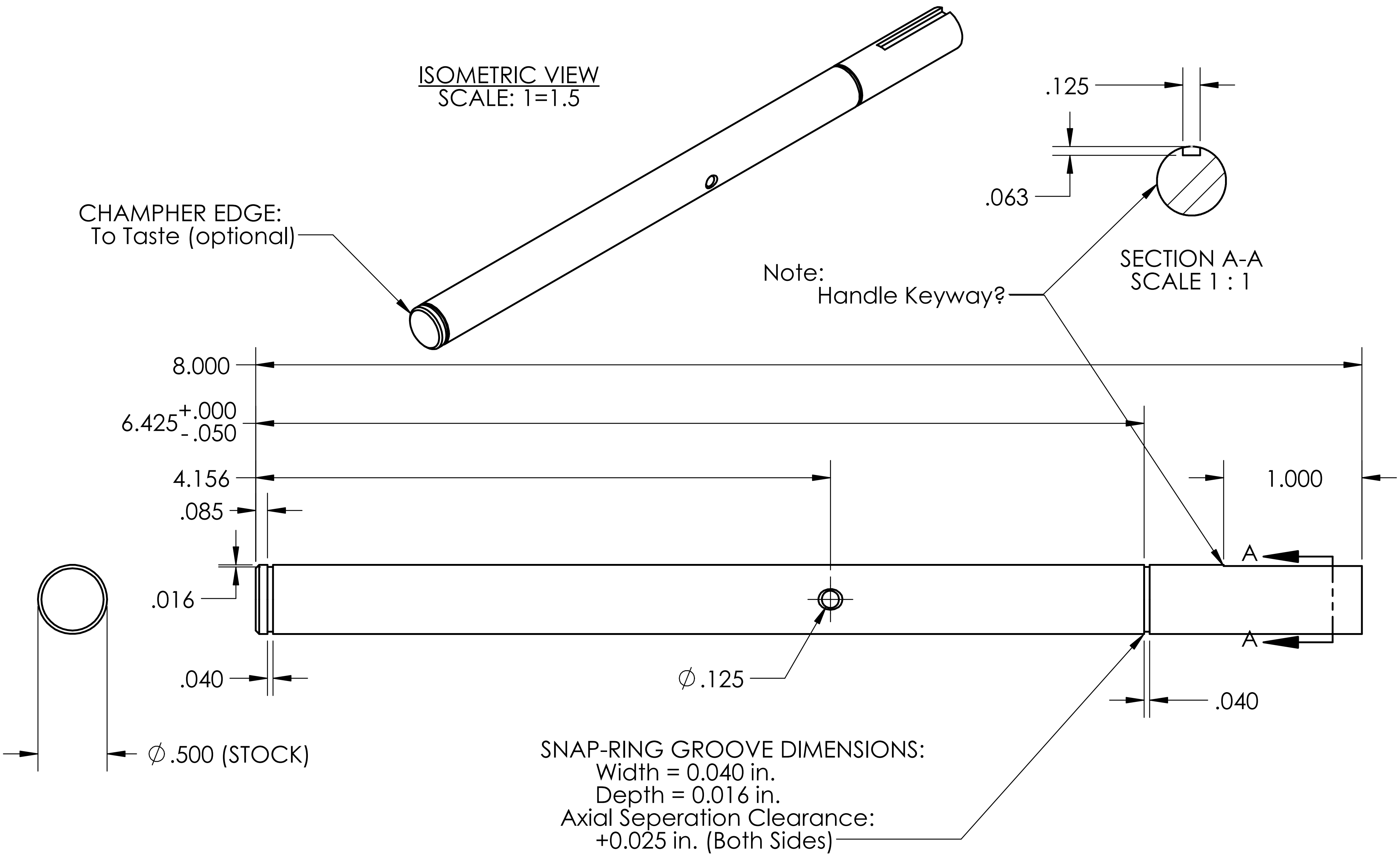

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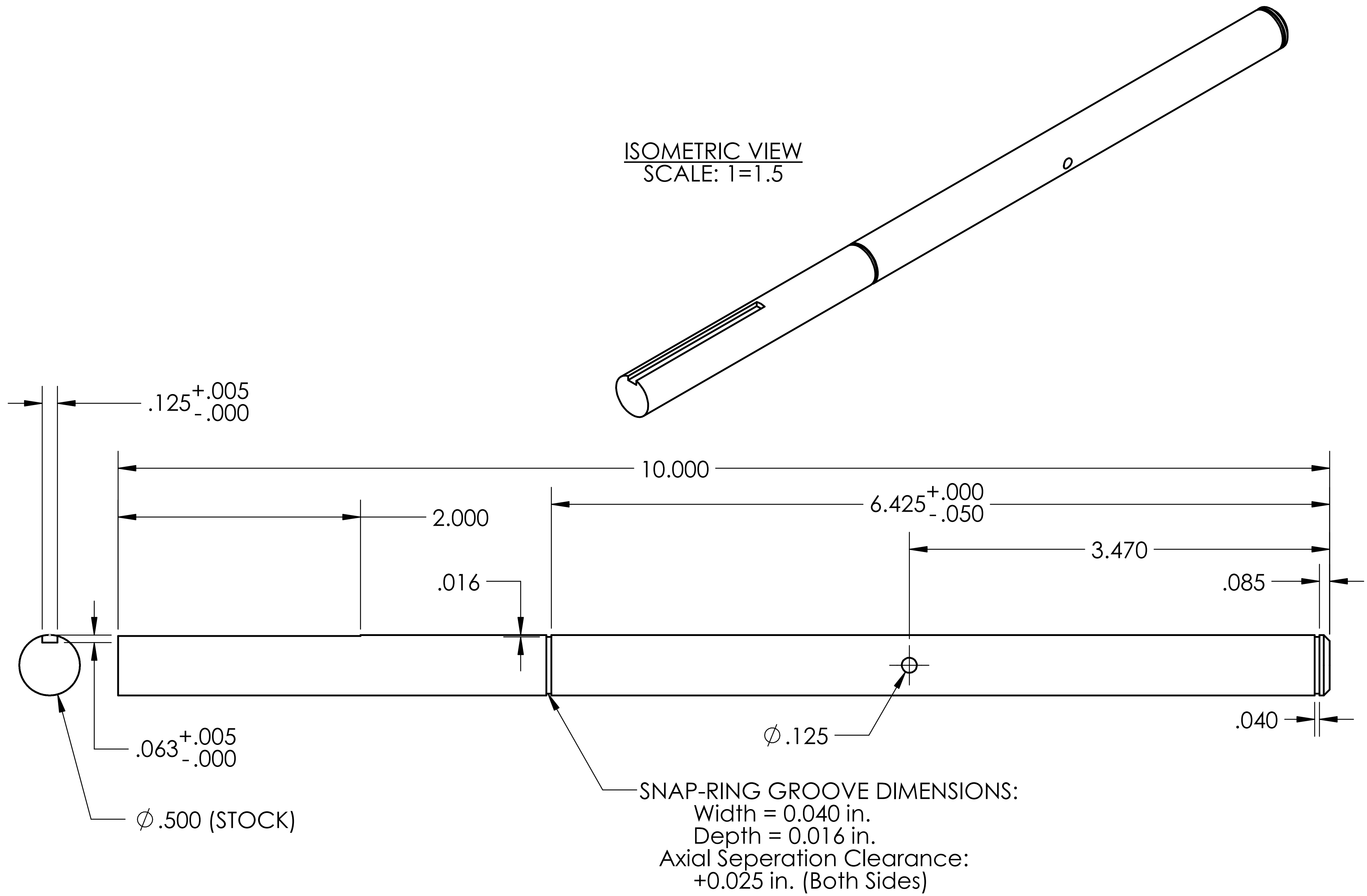

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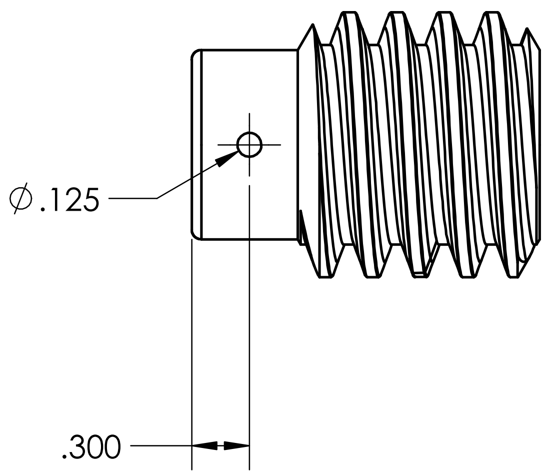

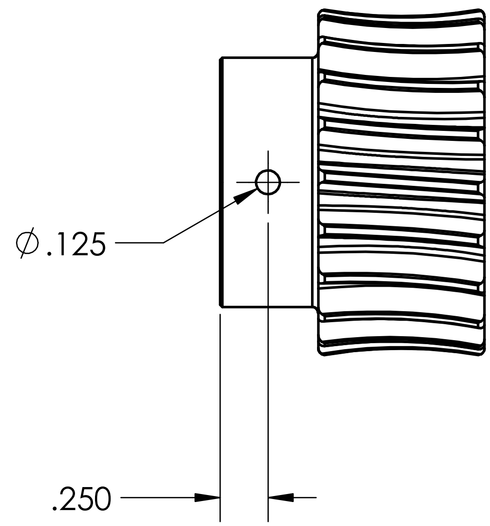

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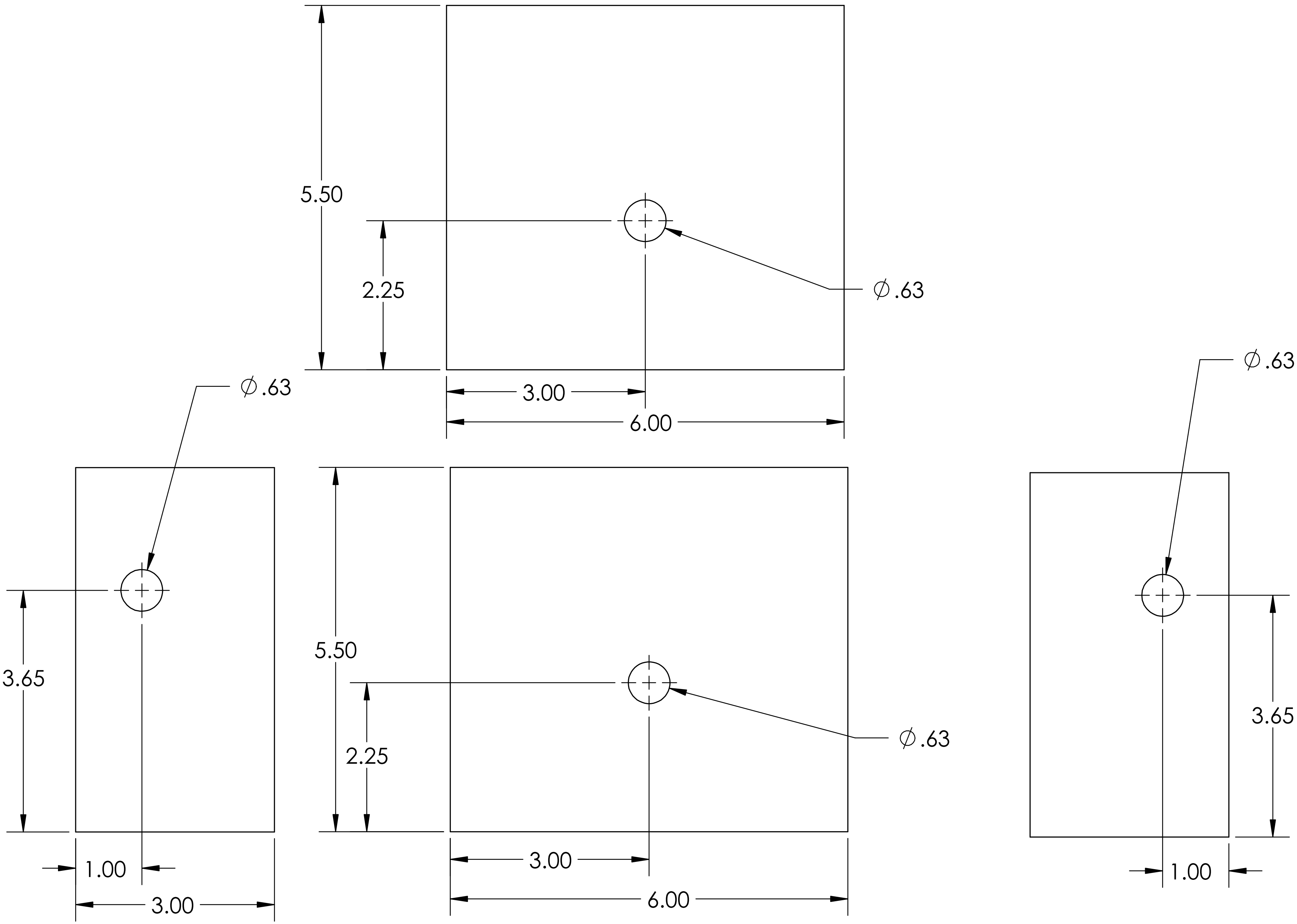

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